VEHICLE DYNAMICS MODEL (VEHDYN II)

1. SCOPE. This document describes the contents and use of a computer model developed by Mobility Systems Division (MSD) used to quantify a vehicle's ride dynamics as it traverses a rigid terrain profile at a constant speed. This two-dimensional vehicle dynamics model, called VEHDYN II, is used to predict the translational motion history of axles, suspension components, the sprung mass center of gravity (CG), and a driver's seat, the angular motion history of suspension beams and the sprung mass, as well as the absorbed power history for a driver's seat. Given this kind of output, ride and shock limiting speed relations can be developed for any given wheeled, tracked, or half-tracked vehicle that can be approximated by the simple suspension types available in VEHDYN II.

2. BACKGROUND: FROM VEHDYN TO VEHDYN II. VEHDYN, which was developed by MSD in 1974, provided ride and shock simulation capability to support AMM. VEHDYN predicted gross motions of a vehicle's chassis as it crossed rough terrain or discrete obstacles, and calculated absorbed power or peak vertical accelerations at a specified vehicle location (eg., a driver's seat). VEHDYN was never intended to be a detailed model for assessing intricate effects of various vehicle components on ride dynamics; rather, it was to be a quick-use tool in which a user could lump gross mass and suspension properties and produce reasonable estimates of ride and shock limiting speed relations.

In 1978, a NATO working group adopted AMM and VEHDYN as the standard reference for evaluating the cross-country mobility performance of vehicles. The AMM with VEHDYN became the earliest version of the NRMM, the Nato Reference Mobility Model. VEHDYN subsequently was cast into widespread usage and changes by different users led to many program inconsistencies and errors, as well as an unwieldy structure. There was also no good user documentation available. These problems, along with a desire to upgrade VEHDYN in many areas, led MSD to develop VEHDYN II in 1986.

3. PROBLEM DESCRIPTION.

3.1 Terrain profile. VEHDYN II requires input of an ordered set of coordinates to describe a rigid terrain profile over which the vehicle will travel. This data is stored in a single input file called the profile data file.

3.2 Vehicle representation.

3.2.1 General. VEHDYN II also requires input data to fully describe a suspended vehicle, including sprung mass center-of-gravity (CG) location, sprung mass moment of inertia, type and locations of suspensions, driver's seat, and wheels, suspension spring and damper tables, and, for a tracked or half-tracked vehicle, track tension and track feeler parameters.

Coordinates of several key locations must be specified to describe the vehicle in a mathematical representation called the zero-force configuration, in which the vehicle is positioned such that each of the suspension springs neither feels nor exerts any force. The zero-force configuration is necessary to define a reference spring displacement equal to zero.

A vehicle representation for which it is easier to obtain data is its settled, or equilibrium, position which depicts a vehicle as it rests under its own weight on level ground. MSD has a preprocessor to VEHDYN II called PREVDYN2 that converts a user-inputted settled vehicle representation and outputs the vehicle's zero-force configuration data required by VEHDYN II.

3.2.2 Suspension types. VEHDYN II currently has four types of suspensions available to the user. They are the unsprung, the independent, the walking beam, and the bogie. An unsprung suspension is just a wheel mounted rigidly from the sprung mass without any springs. All four suspensions are shown in
Figure 1.

The independent suspension (Figure 1b) contains a single spring-damper combination with a wheel at one end. It has one degree of freedom (dof) and is assumed to remain perpendicular to the frame underside along its entire length with no flexure.

The walking beam suspension (Figure 1c) has 2 dof's and is made up of a spring-damper combination rigidly connected in a perpendicular fashion to the frame underside and a rigid beam with a wheel at each end. The beam is free to rotate about a pivot point (Point C in Figure 1c), but can be restricted in its angular movement by a frictional rotational damper and a rotational bump stop. The walking beam also has optional outboard dampers (Lines EG and FH in Figure 1c) which can be attached anywhere along the beam and connected anywhere to the frame underside.

The bogie suspension (Figure 1d) has 3 dof's and is the most complex suspension in VEHDYN II. The beam (Line DE in Figure 1d) is free to rotate about pivot point C. As with the walking beam, this beam can be optionally restricted via both a frictional rotational damper and a rotational bump stop. At each end of the beam, a spring is rigidly fastened in a perpendicular fashion and has a wheel at its end. A damper can be added that connects from each wheel to the frame underside.

3.2.3 Spring and damper models. One of the significant improvements in VEHDYN II is in the way springs and dampers can be modeled. In VEHDYN, a single curve was used to model both the loading and the unloading behavior. In VEHDYN II, hysteretic effects can be modeled in both springs and dampers by inputting separate loading and unloading relationships as well as a transition exponential curve to allow movement between loading and unloading curves. Figure 2 displays the general cases of spring force-deflection and damper force-velocity relationships.

Furthermore, a damper can be modeled to reflect a displacement dependency using tables, called damper modification coefficient tables, of multiplicative coefficients that are functions of displacement that can be used to modify the force that comes from the nonlinear hysteretic force-velocity data described above.

3.2.4 Wheeled-vehicle interactions. Wheeled vehicles interact with the rigid terrain profile using two models. The first model quantifies the forces being imparted to the vehicle because of tire deflections produced as the vehicle negotiates the ups-and-downs of the rigid profile. This model is called the continuous spring model and is portrayed in Figure 3. For each wheel, the user must input a value of deflection and corresponding vertical force which represents one point from a tire's force-deflection relationship. The model generates forces due to any and all other values of tire deflection. At any given point along the terrain, the continuous spring model produces vertical and horizontal force components acting at each wheel's center, as well as an applied moment for drive wheels.

VEHDYN II uses a second model to allow the user to specify which wheels are to be designated as drive wheels and which are to be towed wheels. The tractive forces required to balance out the horizontal forces coming from all vehicle-terrain interactions are distributed among the drive wheels. The tractive force for each drive wheel is assumed to be proportional to the continuous spring model's normal force component as shown in Figure 4.

3.2.5 Tracked and half-tracked vehicle interactions. Tracked and half-tracked vehicles use the same interaction models as described for wheeled vehicles as well as three additional models. The first model is the track tension model shown in Figure 5. The model consists of vertical interconnecting springs between all wheels enclosed by the track envelope. The forces computed are such that they tend to realign the wheels contained by the track. This means that if a wheel moves upward or downward relative to its neighbors, the tension model tends to pull it back up to the level of its neighbors. Required input is a linear spring constant for each spring; there must be one spring between each pair of wheel neighbors contained by the track envelope.

The second interaction model for tracked vehicles involves contact of either spridler (i.e., the idler or the sprocket) with the terrain profile. This interaction is handled exactly as wheel-terrain contact using the continuous spring model.

The third interaction model for tracked vehicles is contact between the terrain profile and either the forward or the rear track feeler. A track feeler is the section of track between one of the spridlers and the nearest corresponding road wheel. Figures 6 and 7 portray the model for the forward feeler and the rear feeler, respectively. The track feeler is modeled by a linear spring (with user-supplied spring constant k_f) perpendicular to the feeler. When the terrain profile intersects the feeler (see Figure 6), the displacement δ_{max} is determined producing a force F_N normal to the feeler. Tension T_f in the feeler is used to balance this normal force F_N and then applied to the spridler and nearest road wheel to additively modify the force and moment components already computed from the continuous spring model.

For a track or half-tracked vehicle, the user-designated drive wheels are used to balance all of the horizontal forces, including those from the continuous spring model and the feeler-terrain model. It is recommended that all of the wheels contained within the track envelope be made drive wheels.

3.2.6 Driver's seat. The final element making up the vehicle representation is that of an optional driver's seat. Options for a driver's seat include no seat, an unsuspended seat, or a seat suspended on a nonhysteretic spring-damper combination. Computations for the driver's seat include vertical accelerations and absorbed power histories, maximum vertical acceleration, and average absorbed power. The motion of the driver's seat is not assumed to affect the overall vehicle dynamics and is not fed back through the sprung mass dynamics calculations. The driver's seat motion is assumed to be purely a result of the motion of the sprung mass.

Absorbed power is a way to quantify the rate at which a human absorbs vibrations. It is computed using the same algorithm as was in VEHDYN which takes filtered vertical accelerations and computes instantaneous (vertical) absorbed power using a digital algorithm that was translated from an analog circuit model. The instantaneous absorbed power is time averaged as the calculation proceeds to obtain average absorbed power.

3.3 Control file. Finally, a control data file is required to input information regarding what problem to run and how the problem is to be run. This file contains descriptors selecting the desired vehicle and profile data sets, vehicle speed, computational time step, maximum time or distance to run, low-pass filter frequency for accelerations feeding the absorbed power algorithm, as well as output increments for each of three output data files.

4. COMPUTATIONAL METHODOLOGY. Figure 8 is a generalized flow chart for VEHDYN II. After the data described previously is read in, the vehicle is settled under its own weight using an iterative technique called the stiffness method; the goal of the stiffness method is to determine how much each suspension spring must compress to support the sprung mass in equilibrium. This process involves initially guessing values for wheel and CG vertical displacements and frame and beam angular displacements. Spring and wheel deflections are then computed and used to obtain appropriate spring constants from the input spring loading curves and wheel model. The stiffness method is applied to obtain new values for the displacements. The process is repeated until the displacements converge.

Once the vehicle's equilibrium position is determined, time, motion variables, and vehicle location along the profile is initialized. Then time is incremented and the vehicle is moved ahead a distance equal to the input vehicle speed multiplied by the time increment. Key vehicle location coordinates are updated. Wheel forces are computed. Then accelerations for the vertical and angular CG motions as well as each individual suspension's dof are computed by applying Newton's second law (i.e., acceleration equals the sum of applied forces divided by mass (for linear accelerations) and angular acceleration equals the sum of applied moments divided by moment of inertia (for angular accelerations)). Necessary driver's seat computations are made. Motion variables are then integrated using a Runge-Kutta-Gill technique to obtain new velocities and displacements for each dof. The process continues until either the maximum time to run or maximum distance along the profile is achieved.

5. OUTPUT. Three output files are generated by VEHDYN II. The first is a formatted time-history file of the entire event. It is a printable file and contains both an echo of the various input parameters and tables as well as step-by-step history of the vehicle's movement down the profile. Coordinates of key vehicle locations, key displacements, velocities, and accelerations of suspension components, as well as wheel forces and moments are included. The output increment

is user-specified in the control data set and should be a multiple of the integration time step.

A second file, called the binary time-history file, contains the same information as the first file (but using its own output increment) in a nonprintable binary format. This file can be and is intended to be used as input to any postprocessing to be accomplished.

Finally, a third file, called the vertical absorbed power table, contains a formatted, printable table summarizing the driver's seat calculations including vertical accelerations (both raw and filtered), vertical absorbed power (instantaneous and time-averaged), and vehicle location along the profile, all tabulated at a constant output increment which can be different from the increments used in the previous two files. As with the other output increments, this output step should also be a multiple of the integration time step.

6. APPLICATIONS. The principal intent of VEHDYN II has always been to predict vehicle ride and shock limiting speeds to characterize vehicle ride and shock performance. This section will begin with a discussion of how VEHDYN II can be used to produce ride performance and shock performance information for a given vehicle. VEHDYN II can also be used to compare measured dynamics data, like accelerations, velocities, and displacements of various vehicle locations and/or components and average absorbed power (measured with WES' ride meter) with a computer simulation. Furthermore, VEHDYN II can be used in the design process of new or modification process of existing vehicles although in a relatively crude way. The upcoming subsections will discuss these different ways of using VEHDYN II.

6.1 Ride performance. Ride is defined as the random semiuniform vibrations transferred by the vehicle to the driver or other occupants as a result of traveling over an uneven surface. One of the principal measures of ride comfort is absorbed power. A criterion of 6 watts of absorbed power has been established as an upper bound of vibration that will permit crew members to effectively perform their tasks. Experimental test results have shown that there is quite often only a small increase in speed at 15 or 20 watts over that at 6 watts, because the 6-watt absorbed power levels usually occur when the vehicle's suspension begin "bottoming out" and producing discrete shock loads. Slight increases in speed beyond this point significantly increase the intensity and frequency of these shock loads, which in turn rapidly increase the absorbed power levels. These high absorbed power conditions are not considered to be an effective or meaningful measure of basic ride characteristics. Consequently, the ride performance of a vehicle is based on the speed at which the 6-watt absorbed power level occurs on test courses that cover an acceptable range of surface roughness.

Surface roughness, measured in inches, is obtained from profiles usually measured at 1-foot intervals with a rod and level. Surface roughness is described in terms of root-mean-square (rms) elevation. Before surface roughness is calculated, the terrain profile is filtered with a special numerical filter to remove the major effects of wavelengths longer than about 60 feet, which do not influence the vehicle's ride responses; a program separate from VEHDYN II is used to determine surface roughness. The results of analyzing the surface roughness distributions of extensive profile data representing countries throughout the world revealed that more than 90 percent of the surface roughness fell below values of about 3.0 inches rms. It is therefore recommended that ride performance simulations use at least four or five profiles ranging in roughness from about 0.5 to 3.0 or 3.5 inches rms to suitably describe surface roughness for ride performance evaluations. This range is valid for both wheeled and tracked vehicles.

The first step in determining ride performance for a specific vehicle is to develop a basic ride quality graph such as that shown in Figure 9. This graph can be developed from either experimental tests or by computer simulations. The particular graph in Figure 9 consists of absorbed power versus vehicle speed plots for a single vehicle on five test courses (profiles), each with a different surface roughness. The five test courses encompass a suitable range of surface roughness conditions from very mild (0.6 inch rms) to very rough (3.3 inches rms). Each data point represents a single VEHDYN II run at the specific speed. Curves are eyeballed through the data to delineate the trend in which average absorbed power varies with speed on each course.

The ride-limiting speeds are then determined from the intersections of the eyeball curves with the 6-watt line. A cross-plot of the 6-watt speeds versus surface roughness yields the desired ride performance plot shown in Figure 10. It should be recognized that absorbed power watt levels other than 6 watts could be used in the same manner as a criterion to develop other performance graphs. User discretion and experience will minimize the number of VEHDYN II runs required on each test course to adequately determine the corresponding 6-watt speeds.

6.2 Shock performance. Shock is defined as a sudden, severe change in vibration transferred from the vehicle to the driver or other occupants as a result of an impact with a discrete obstacle such as a boulder, log, rice paddy dike, or ditch. One of the principal measures of shock tolerance is the maximum peak acceleration. A criterion of 2.5 g's vertical acceleration has been established as an upper bound of shock. This limit was determined through research and experimental testing at the WES in the early 1960's. In several field test programs, a group of WES drivers was instructed to drive vehicles at the fastest speeds they could tolerate over rough terrain that contained naturally distributed discrete obstacles. Evaluation of the resulting acceleration histories from accelerometers fastened to the chassis directly beneath the driver's seat revealed that the peak acceleration from encountering obstacles seldom exceeded 2.5 g's. That is, the driver routinely adjusted the vehicle speed to levels that prevented peak accelerations of more than about 2.5 g's. Consequently, this driver-imposed limit was chosen as the shock limit criterion. It has been determined that the principal measure of the characteristics of discrete obstacles is the height. Therefore, the shock performance of a vehicle is based on the speed at which a 2.5-g peak acceleration occurs over a group of discrete obstacles that cover an acceptable range of obstacle heights. The range of heights depends on the vehicle and user discretion. For example, the 2.5-g speed for an M60 tank (tracked) crossing a 16-inch obstacle is about 6 or 7 mph while a 6-inch obstacle never produces a 2.5-g level at any speed. However, an M151 jeep (wheeled) cannot cross obstacles with heights greater than its clearance, which is between 12 and 13 inches. The 2.5-g speed for an M151 jeep crossing a 6-inch obstacle is about 10 mph and a 4-inch obstacle will not produce a 2.5-g level. A rule-of-thumb recommendation for obstacle performance simulations is to select a group of half-round obstacles at 2-inch intervals of height ranging from:

6 to 20 inches (tracked vehicles)
4 inches to minimum clearance (wheeled vehicles)

The first step in determining the shock performance for a specific vehicle is to develop a basic shock quality graph such as that shown in Figure 11. This graph can be developed either from experimental tests or by computer simulations. The particular graph in Figure 11 consists of the largest positive or negative peak acceleration (g) versus vehicle speed data plots for a single vehicle crossing five semicircular obstacles. The obstacles ranged in height from 8 to 16 inches. Each data point represents a single VEHDYN II run at the specified speed. Curves were eyeballed through the data to delineate the trend in which the maximum peak acceleration varies with speed over each obstacle. The shock-limiting speeds are then determined from the intersection of the eyeball curves with the 2.5-g line. A cross-plot of the 2.5-g speeds versus obstacle height yields the desired shock performance plot shown in Figure 12.

6.3 <u>Comparing dynamics of existing vehicles with field tests</u>. Motion parameters other than absorbed power and peak accelerations can also be field measured, then compared with output from a VEHDYN II simulation. WES instrumented an M923 5-ton truck (Figure 13) by placing a pitch gyroscope at the CG (to measure sprung mass fore-aft pitching), a string potentiometer (also called a "yo-yo") mounted from the forward axle vertically to the vehicle frame (to measure relative displacement between that axle and the sprung mass), and another yo-yo mounted from the forward axle of the rear bogie vertically to the vehicle frame. Figures 14, 15, and 16 compare the gyro's measured pitch, the forward yo-yo's measured displacement, and the rear yo-yo's measured displacement histories, respectively, for the M923 traversing a 12-inch half-round obstacle at 6 mph with output from VEHDYN II. The binary output file generated by a VEHDYN II run can easily be used as input to any user-developed postprocessor to plot any dynamic quantity that can be measured.

6.4 <u>Design of new or modification of existing vehicles</u>. VEHDYN II can also be used as a first-line design tool in the development of new or modification of existing vehicles. Of course, a more sophisticated tool should be used to finalize a design, but VEHDYN II can be and has been used as a design-elimination resource.

One example is the case of modification of an M60 battle tank to have a suspended mine-clearing roller mounted to its front end. The M60 is a tracked vehicle with 6 independently-suspended road wheels on each side and weighs 112,000 lbs. Figure 17 shows the effect (from VEHDYN II simulations) of adding a 19,000-lb mine roller to the front end, then running the vehicle at 8 mph over a 500-ft test course having an rms surface roughness of 2.4 inches. The top half of the figure shows the vehicle with stock suspensions 16.4 seconds down the course, while the bottom half of the figure shows the vehicle (at the same point) with suspensions #1 (front-most) and #6 (rear-most) having 60-pct stiffer dampers. Notice that the stiffer dampers keep the roller from impacting the ground as they help reduce fore-to-aft pitching of the vehicle. It is much cheaper to examine the effects of various suspension modifications with a computer model like VEHDYN II than it is to actually build a modified vehicle and field test it.

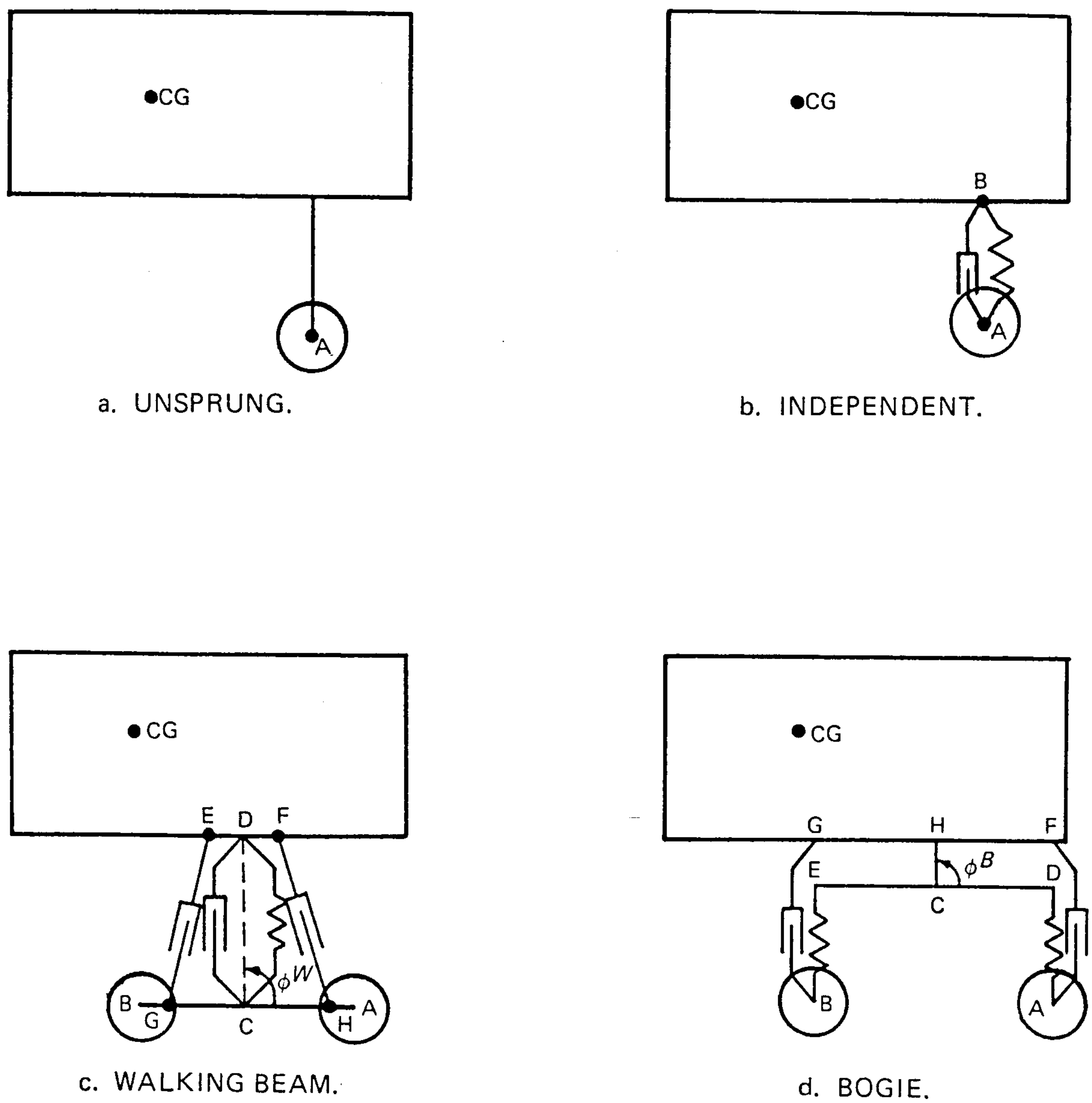
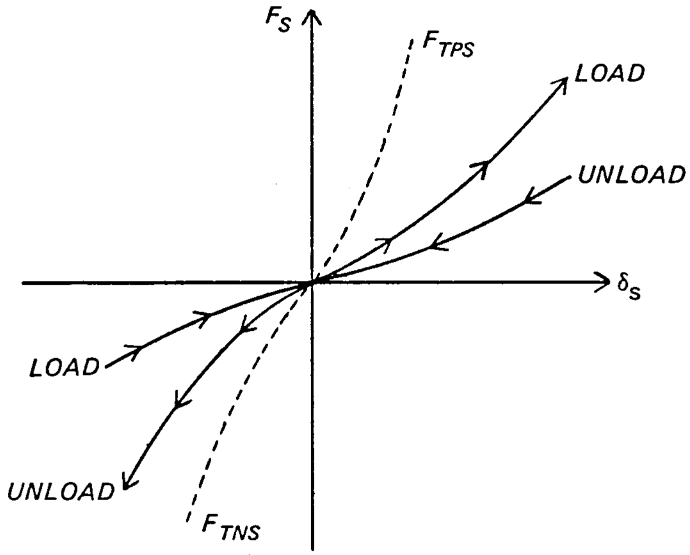


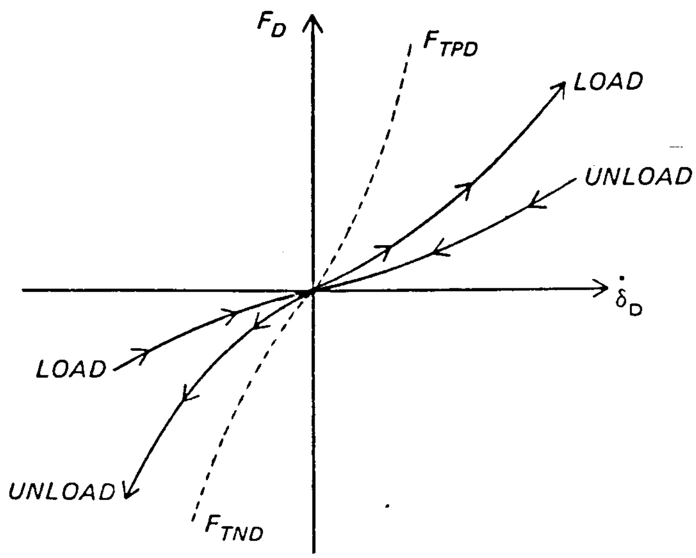
Figure 1. Four basic suspensions modeled in VEHDYN II.



TRANSITION CURVES

$$F_{TPS} = A1\,(e^{B1 \cdot \delta_S} - 1),\ \delta_S > 0$$

$$F_{TNS} = A2(1 - e^{-B2 \cdot \delta_S}),\ \delta_S < 0$$

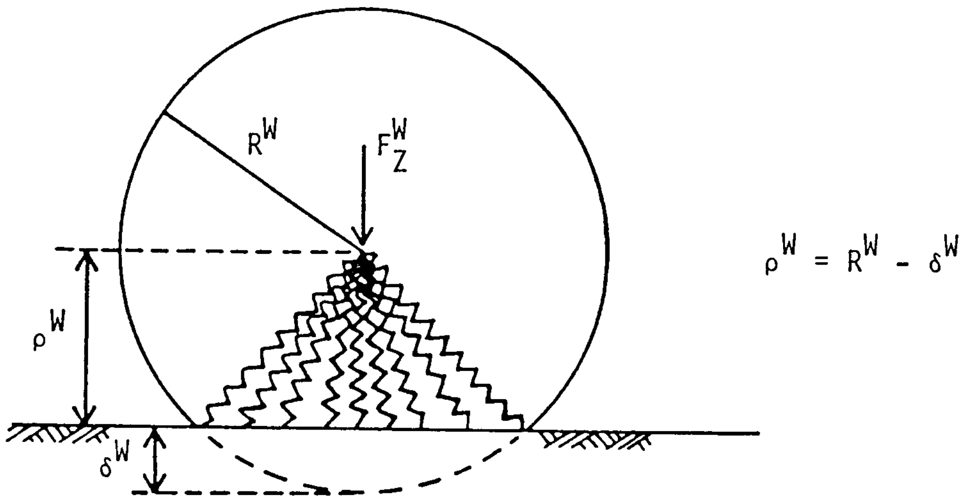


TRANSITION CURVES

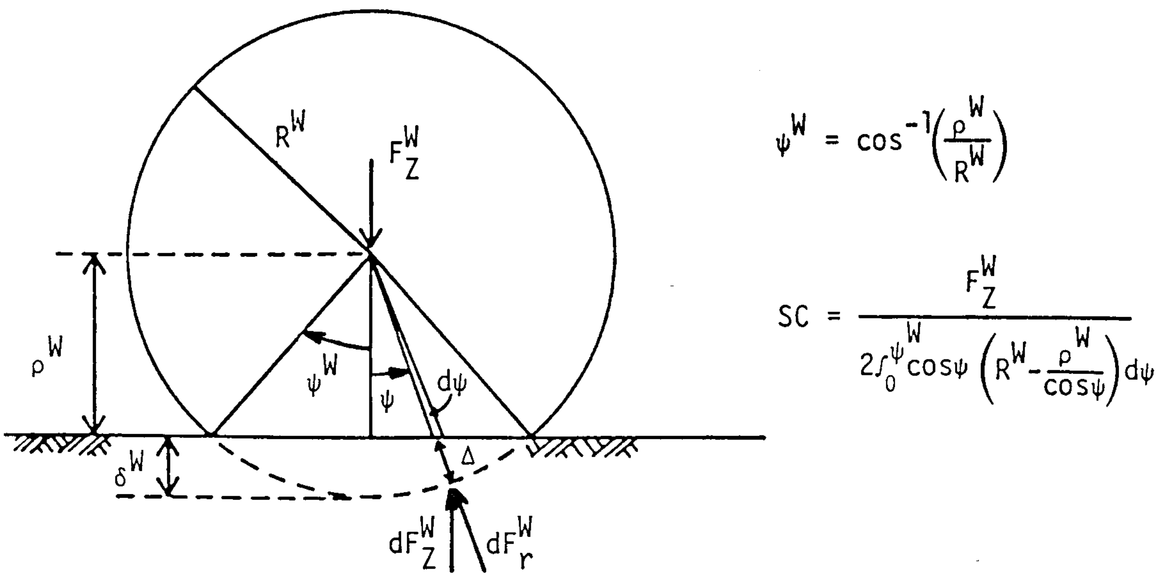
$$F_{TPD} = A3(e^{B3 \cdot \dot{\delta}_D} - 1),\ \dot{\delta}_D > 0$$

$$F_{TND} = A4(1 - e^{-B4 \cdot \dot{\delta}_D}),\ \dot{\delta}_D < 0$$

Figure 2. Nonlinear hysteretic spring force-deflection and damper force-velocity models.

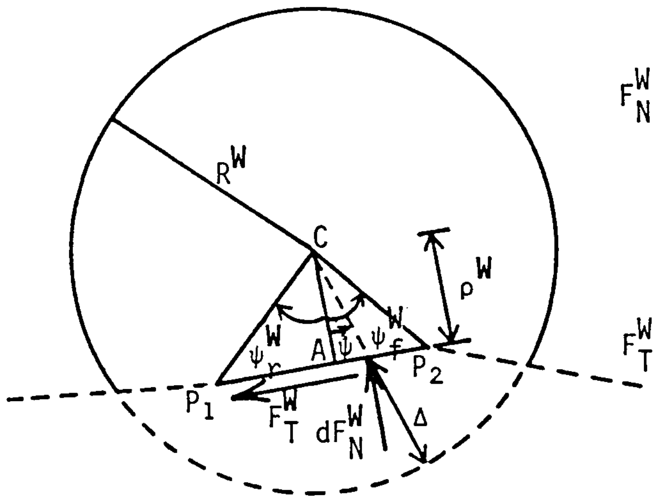


a. Radial springs compressed under load F_Z^W.



b. Determination of radial spring constant SC.

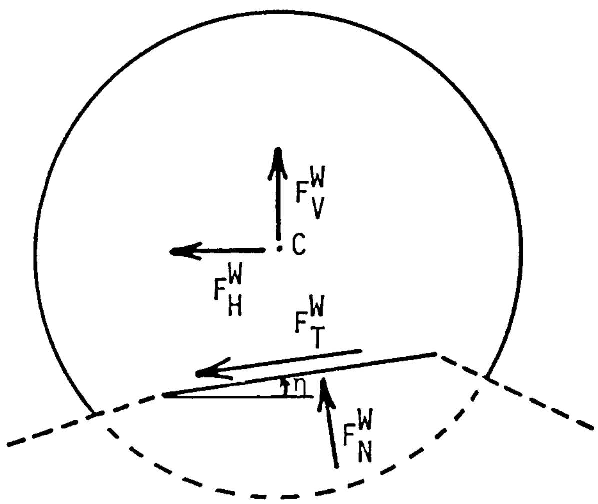
Figure 3. Continuous Spring Model (Sheet 1 of 2).



$$F_N^W = \int_{\psi_r^W}^{\psi_f^W} dF_N^W = SC\,[R^W(\sin\psi_f^W - \sin\psi_r^W) - \rho^W(\psi_f^W - \psi_r^W)]$$

$$F_T^W = \int_{\psi_r^W}^{\psi_f^W} dF_N^W \tan\psi = SC\,\left[R^W(\cos\psi_r^W - \cos\psi_f^W) + \rho^W \ln\left(\frac{\cos\psi_f^W}{\cos\psi_r^W}\right)\right]$$

c. Normal and tangential forces due to interaction between a wheel and a terrain profile segment.

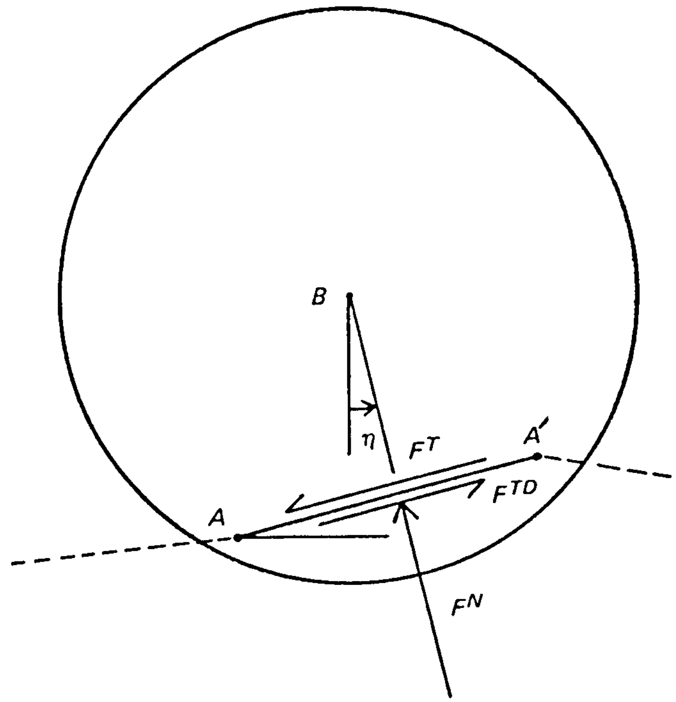


$$F_H^W = \sum_{\substack{\text{all}\\ \text{segments}}} (F_N^W \sin\eta + F_T^W \cos\eta)$$

$$F_V^W = \sum_{\substack{\text{all}\\ \text{segments}}} (F_N^W \cos\eta - F_T^W \sin\eta)$$

d. Resultant horizontal and vertical forces acting through the wheel's center.

Figure 3. (Sheet 2 of 2).



ASSUMPTION:

$F_i^{TD} = C \cdot F_i^N$

[SAME C FOR ALL DRIVE WHEELS AT A GIVEN INSTANT]

$$\Rightarrow C = \frac{F^H_{TOTAL}}{\sum_{i=1}^{NDW} F_i^N \cos \eta_i}$$

WHERE i = SUBSCRIPT FOR THE NDW DRIVE WHEELS ON THE VEHICLE

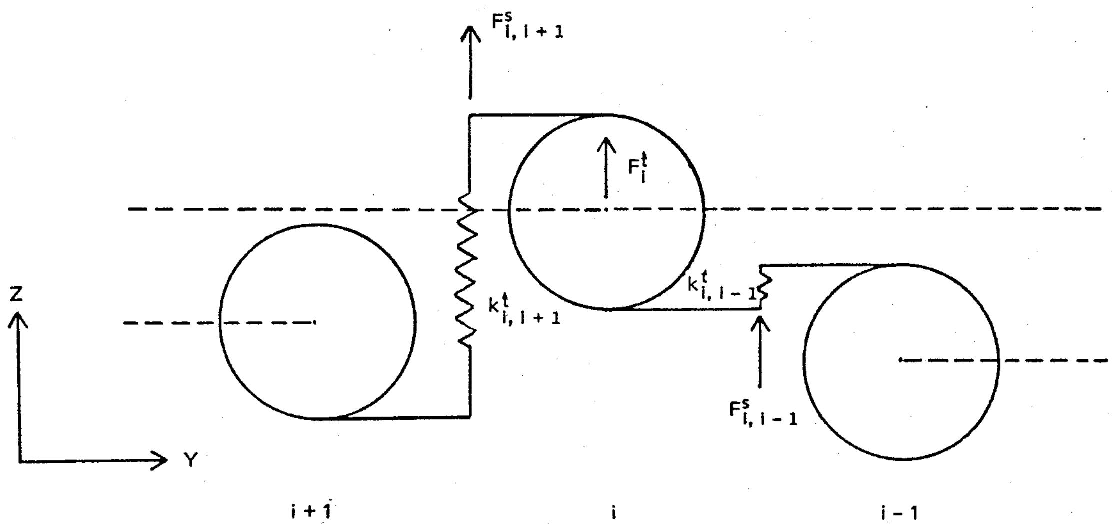
F^T = TANGENTIAL FORCE EXERTED ON WHEEL BY PROFILE AS COMPUTED FROM CONTINUOUS SPRING MODEL.

F^N = NORMAL FORCE EXERTED ON WHEEL BY PROFILE AS COMPUTED FROM CONTINUOUS SPRING MODEL.

F^{TD} = NECESSARY DRIVE-WHEEL TANGENTIAL FORCE TO BALANCE THE HORIZONTAL FORCES.

F^H_{TOTAL} = TOTAL HORIZONTAL FORCES EXERTED ON THE VEHICLE BY THE PROFILE.

Figure 4. Drive wheel forces resulting from interaction with the terrain profile.



ξ_i = VERTICAL DISPLACEMENT OF i^{th} WHEEL'S CENTER

$F^s_{i,j}$ = VERTICAL FORCE EXERTED ON i^{th} WHEEL DUE TO TRACK TENSION SPRING CONNECTING THE i^{th} AND j^{th} WHEELS.

$k^t_{i,j}$ = SPRING CONSTANT FOR TRACK TENSION SPRING CONNECTING THE i^{th} AND j^{th} WHEELS

$F^s_{i,j} = k^t_{i,j}\ (\xi_j - \xi_i)$

$j = \left\{\begin{matrix} i+1 \\ i-1 \end{matrix}\right.$ = GENERIC WHEEL INDEX FOR WHEELS THAT ARE NEIGHBORS OF THE i^{th} WHEEL

F^t_i = TOTAL VERTICAL FORCE CONTRIBUTION FROM TRACK TENSION EFFECT

$F^t_i = F^s_{i,i-1} + F^s_{i,i+1}$

Figure 5. Track tension model for tracked road wheel.

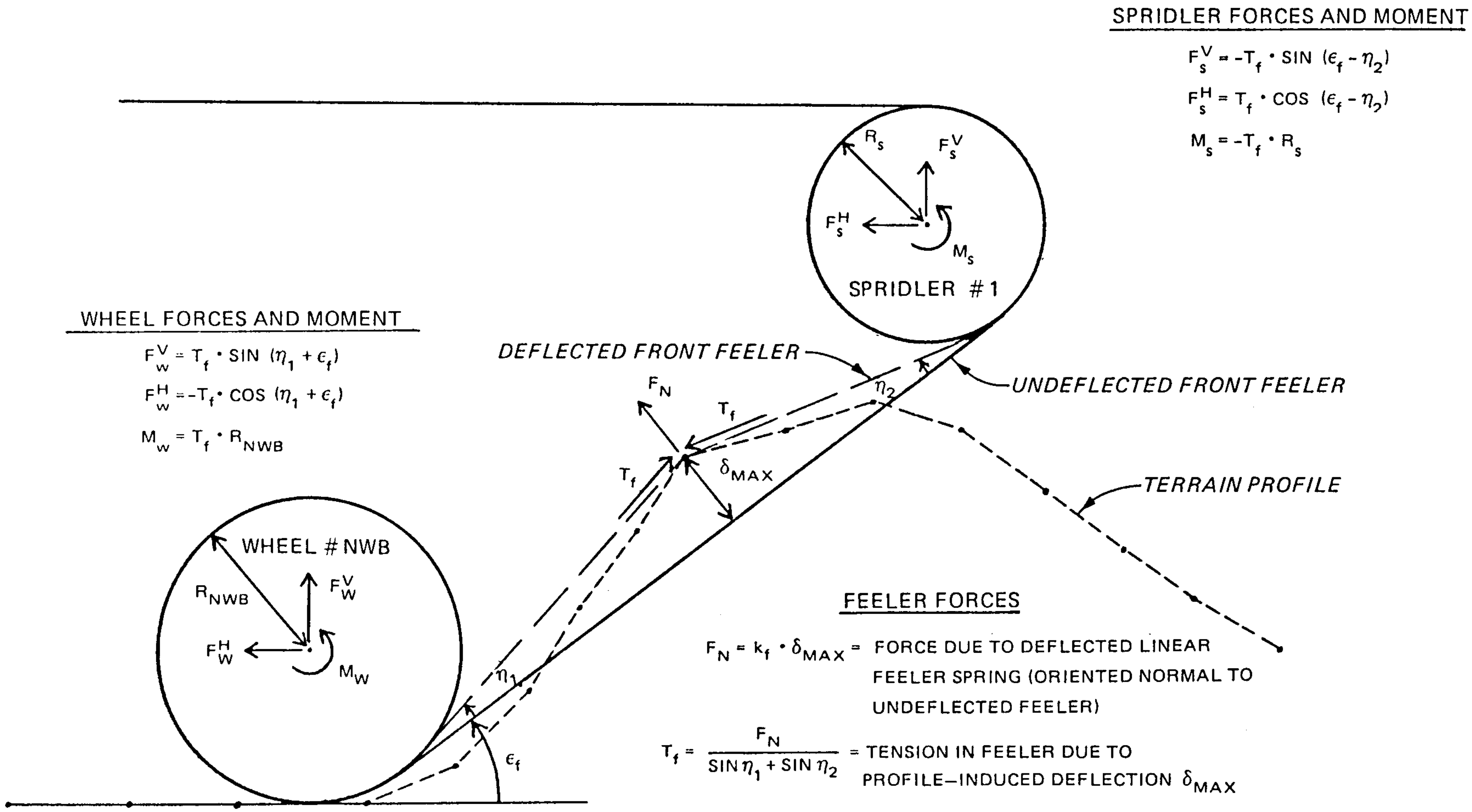


Figure 6. Forward feeler interaction with terrain profile.

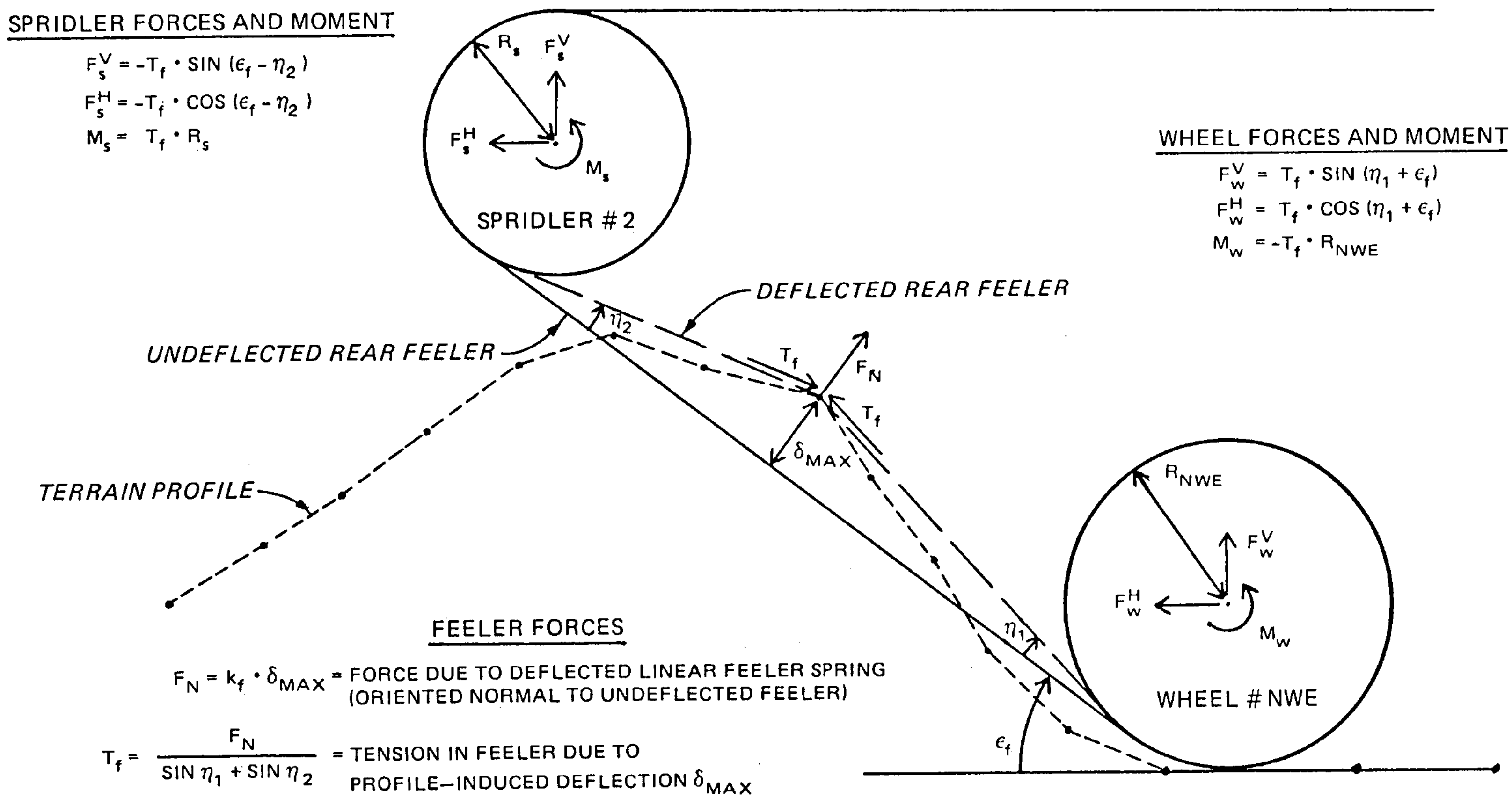


Figure 7. Rear feeler interaction with terrain profile.

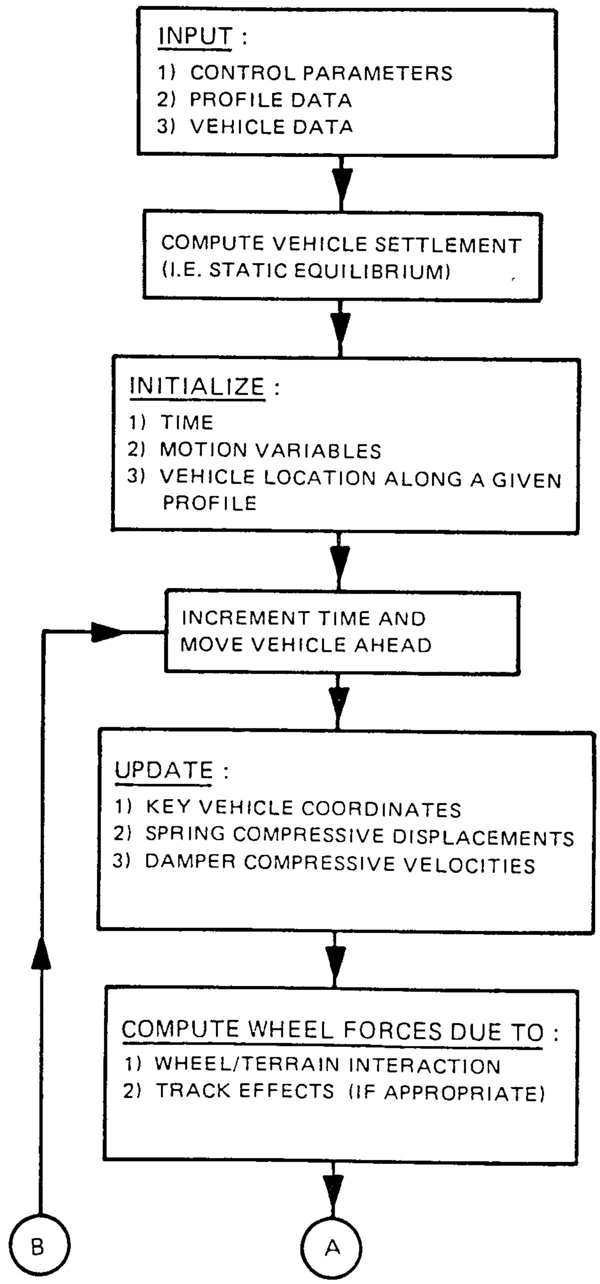


Figure 8. Computational flow chart for VEHDYN II (Sheet 1 of 2).

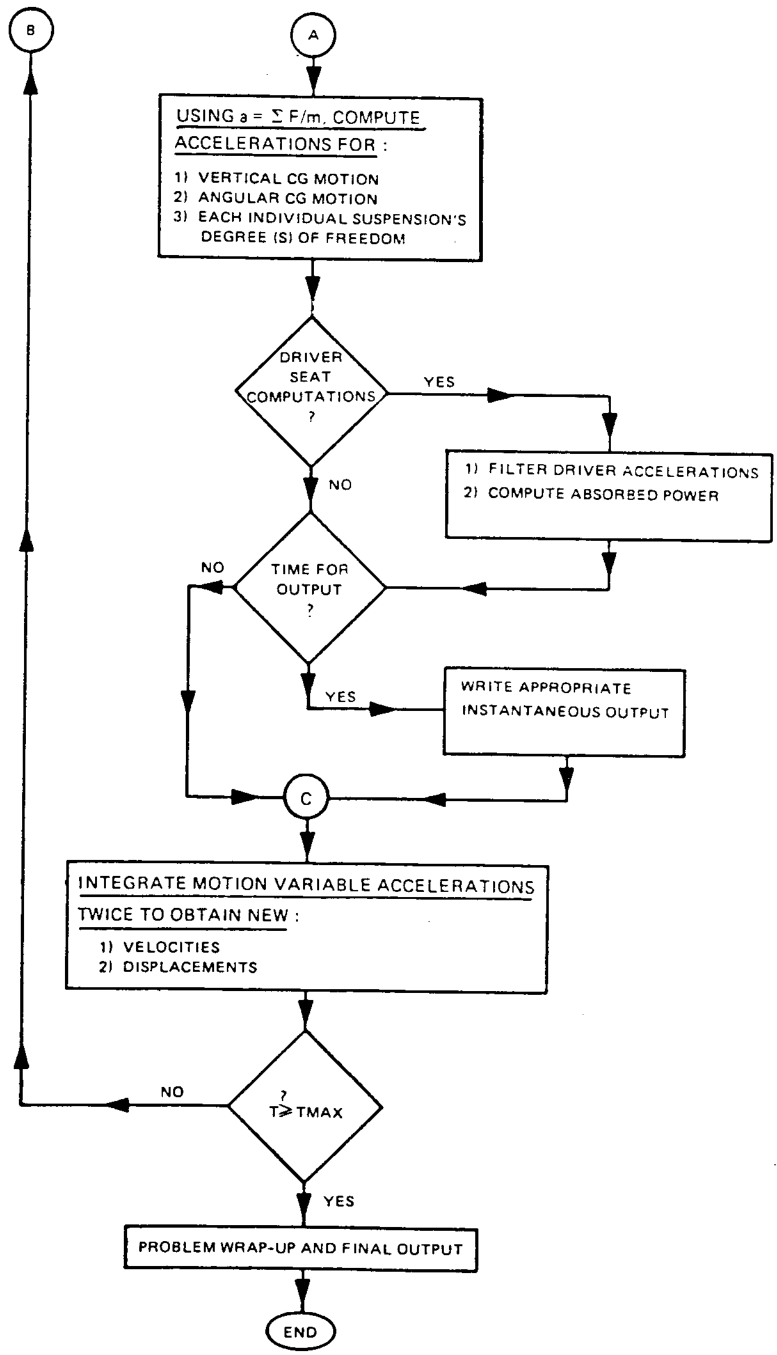


Figure 8. (Sheet 2 of 2).

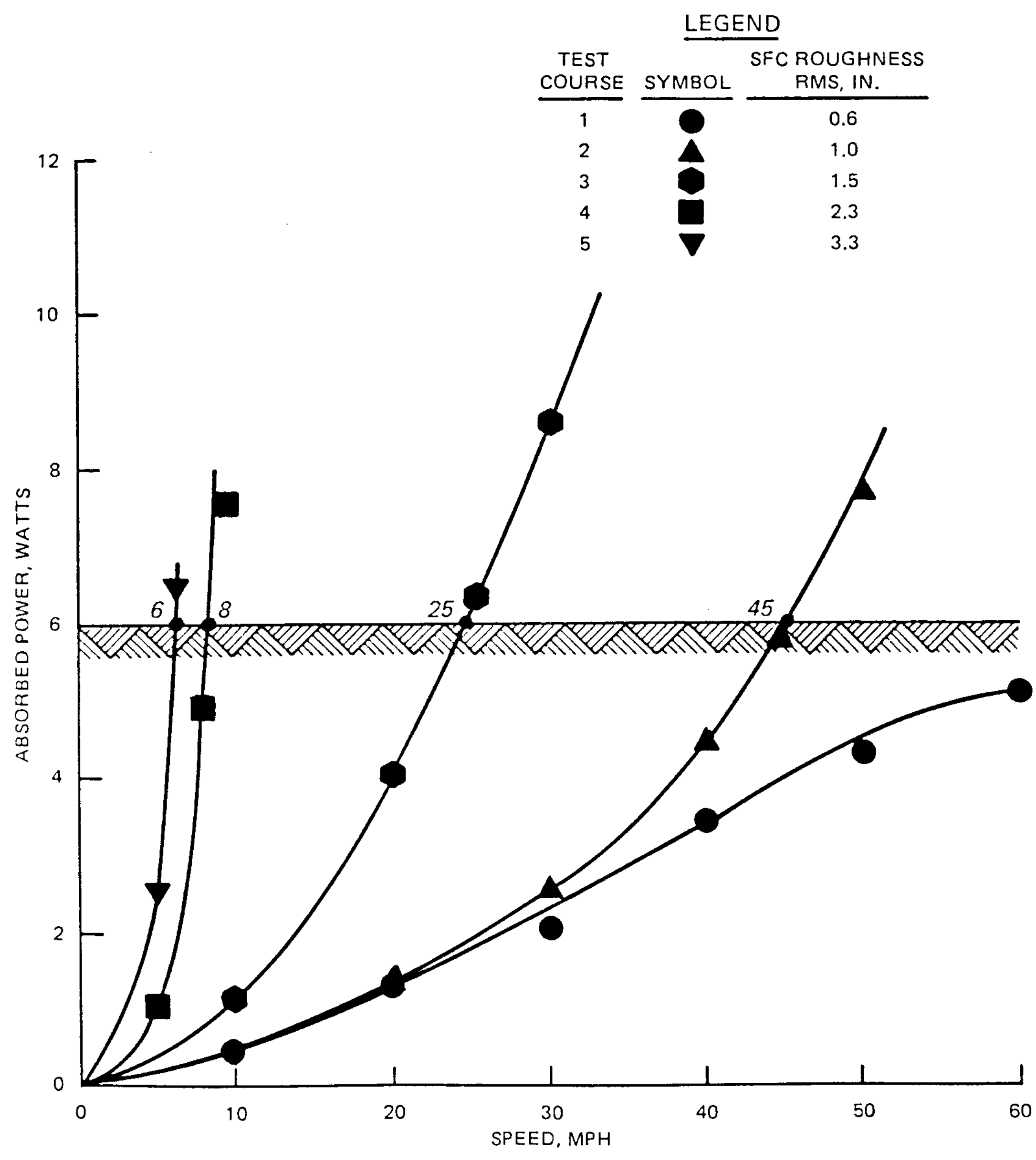


Figure 9. Basic ride quality graph for a hypothetical tracked vehicle.

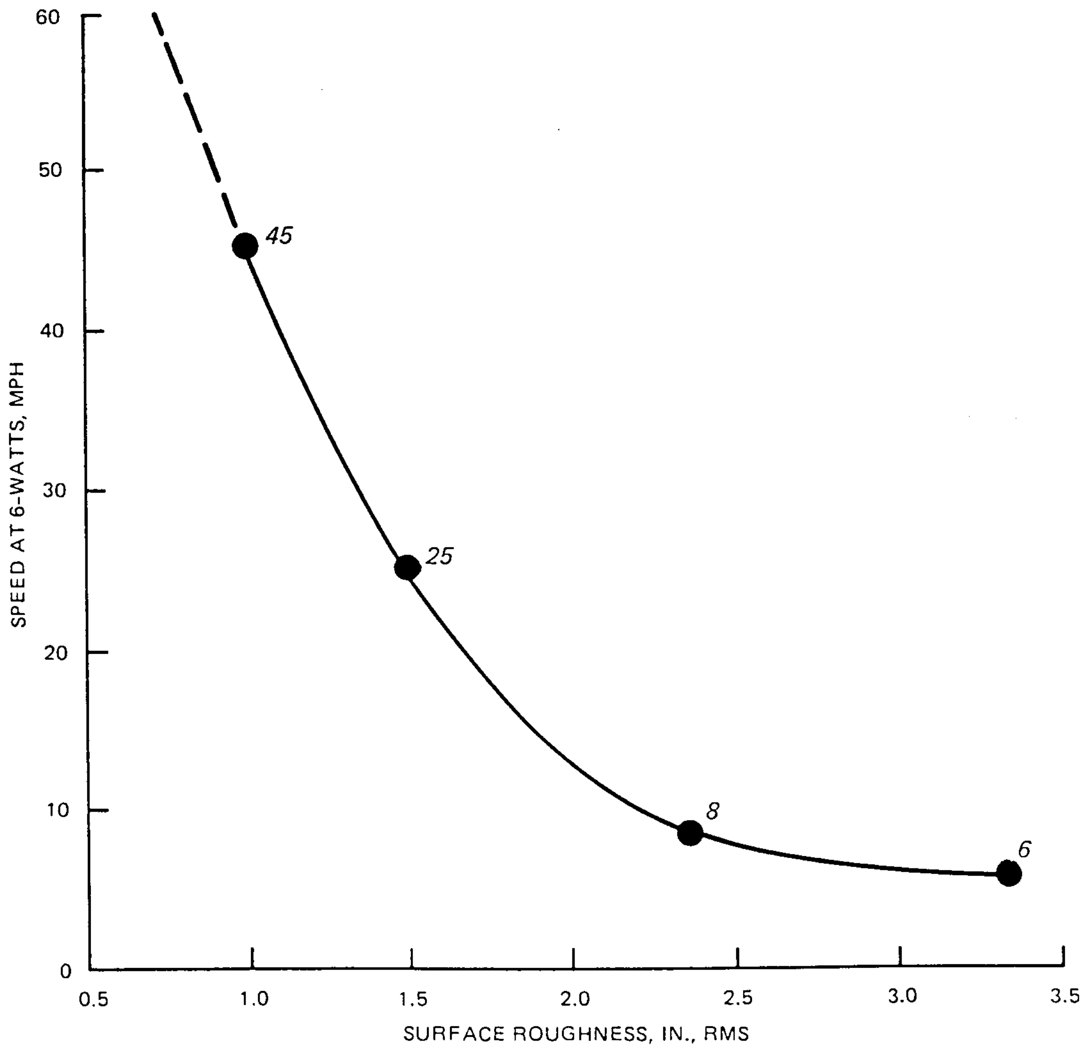


Figure 10. Basic ride performance graph for a hypothetical tracked vehicle.

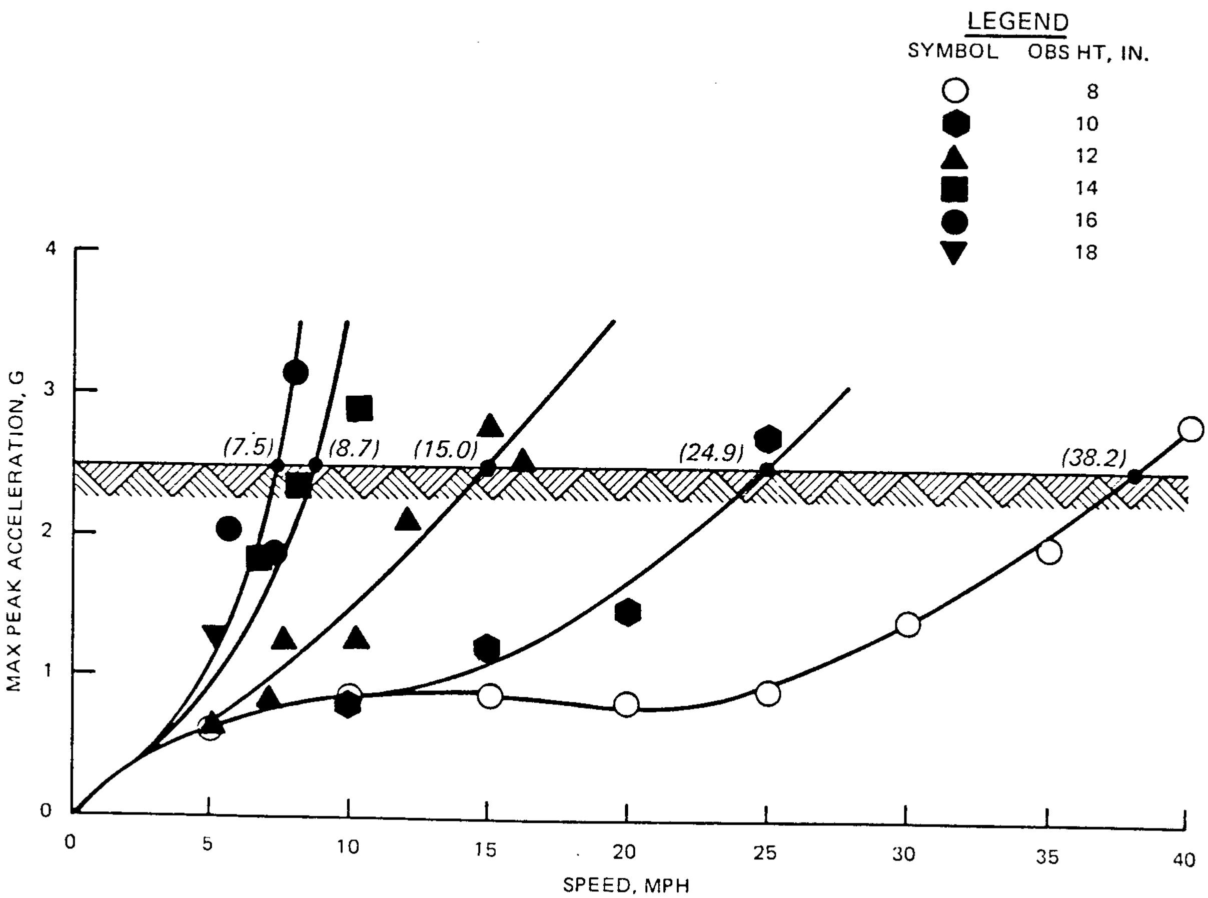


Figure 11. Basic shock quality graph for a hypothetical tracked vehicle.

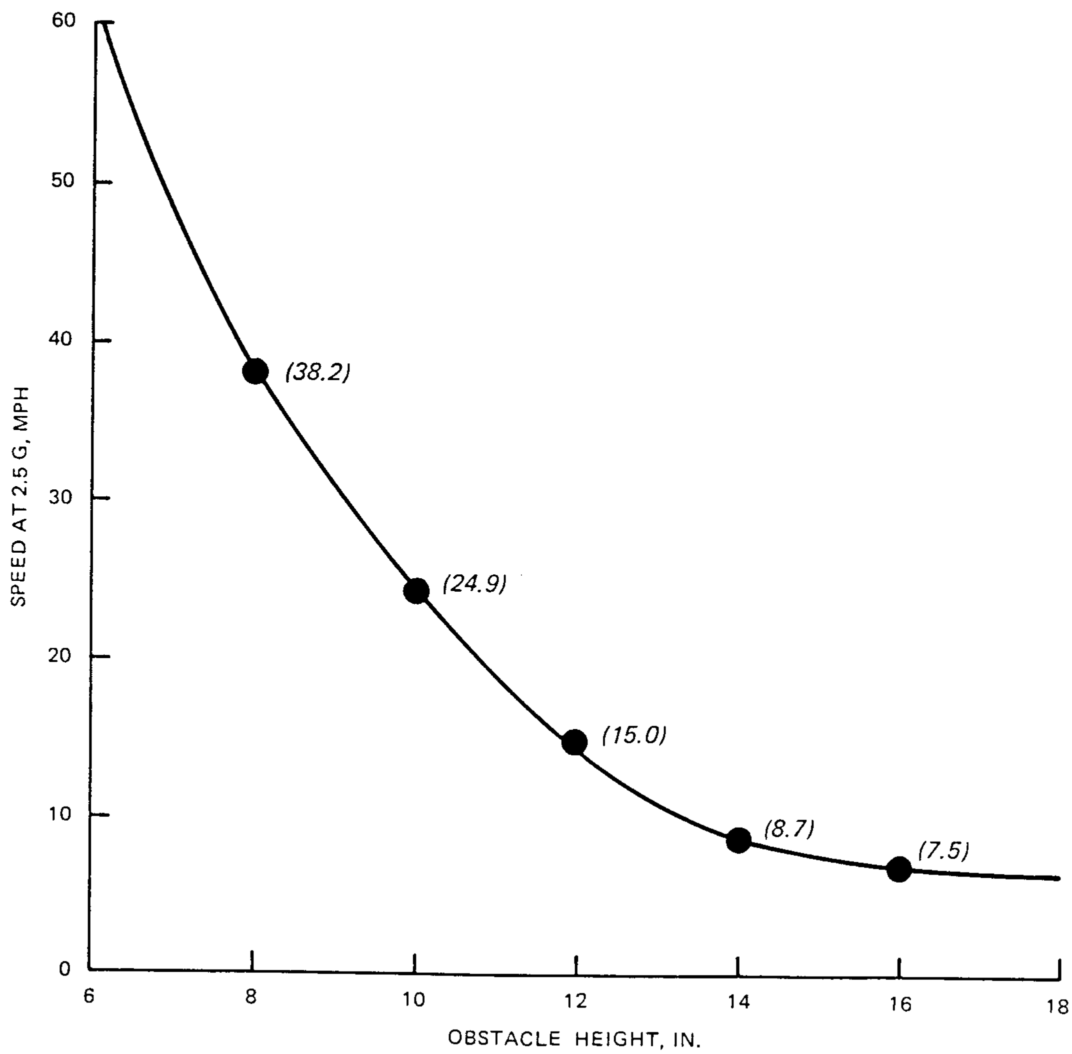
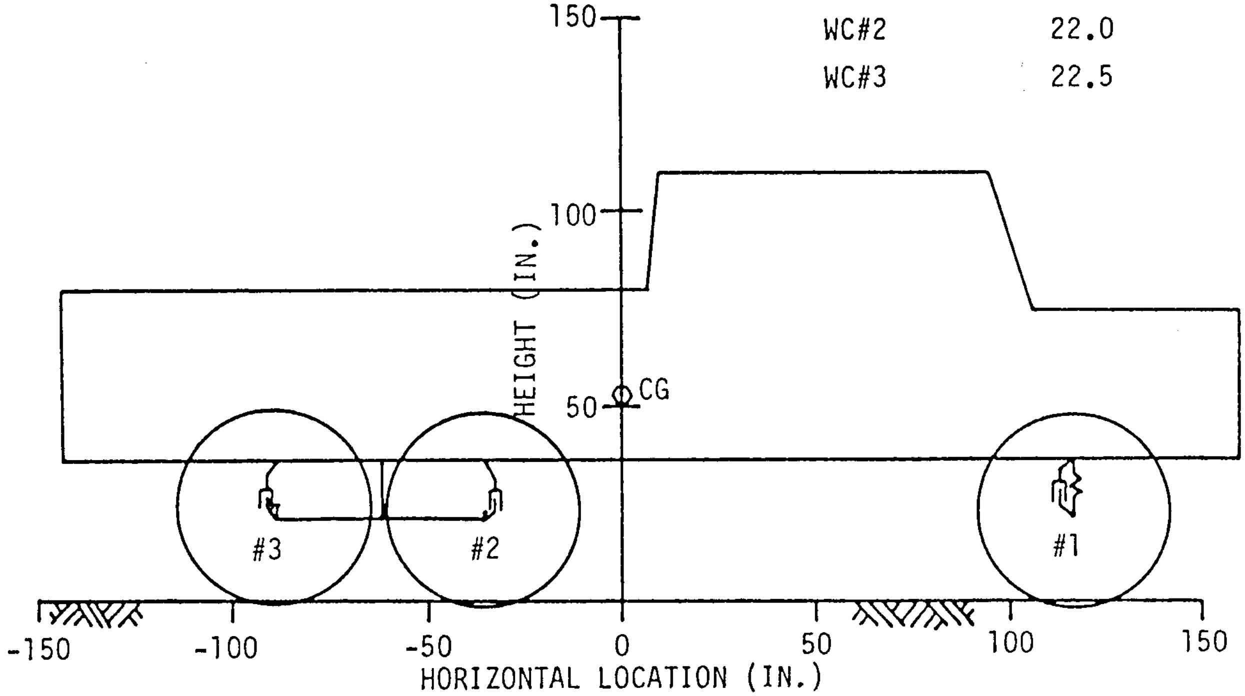


Figure 12. Basic shock performance graph for a hypothetical tracked vehicle.



LOCATION	SETTLED HEIGHTS (IN)	MEASURED LOADS UNDER WHEEL (LB)
CG	53.0	—
WC#1	21.5	5642
WC#2	22.0	5432
WC#3	22.5	5300

Figure 13. Settled configuration for the M923 5-ton truck.

M923 5-TON TRUCK OVER 12-INCH OBSTACLE

V= 6 MPH

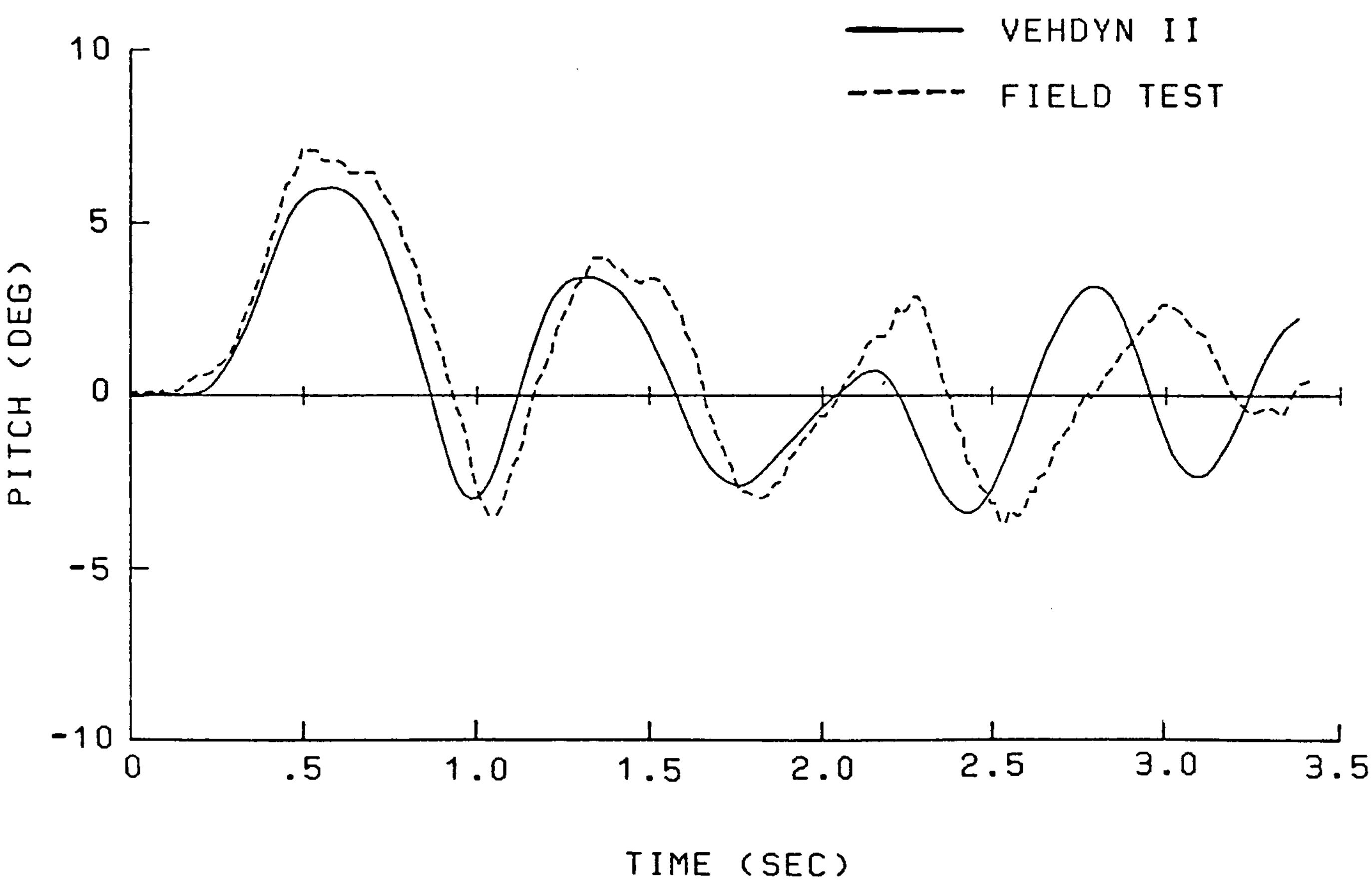


Figure 14. Comparison of measured and calculated pitch histories for the M923 5-ton truck negotiating a 12-inch half-round obstacle at 6 mph.

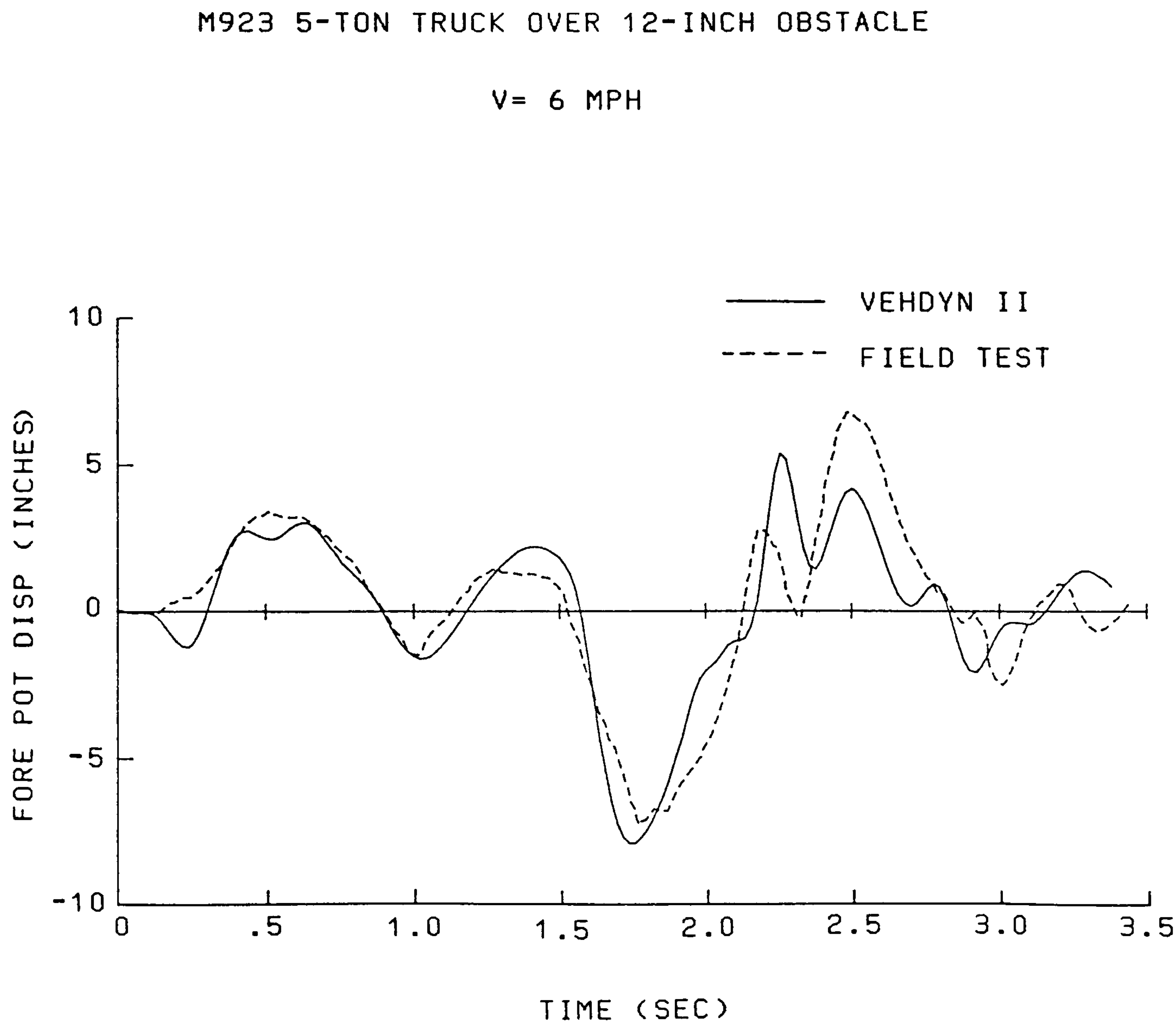


Figure 15. Comparison of measured and calculated forward yo-yo displacements for the M923 5-ton truck negotiating a 12-inch half-round obstacle at 6 mph.

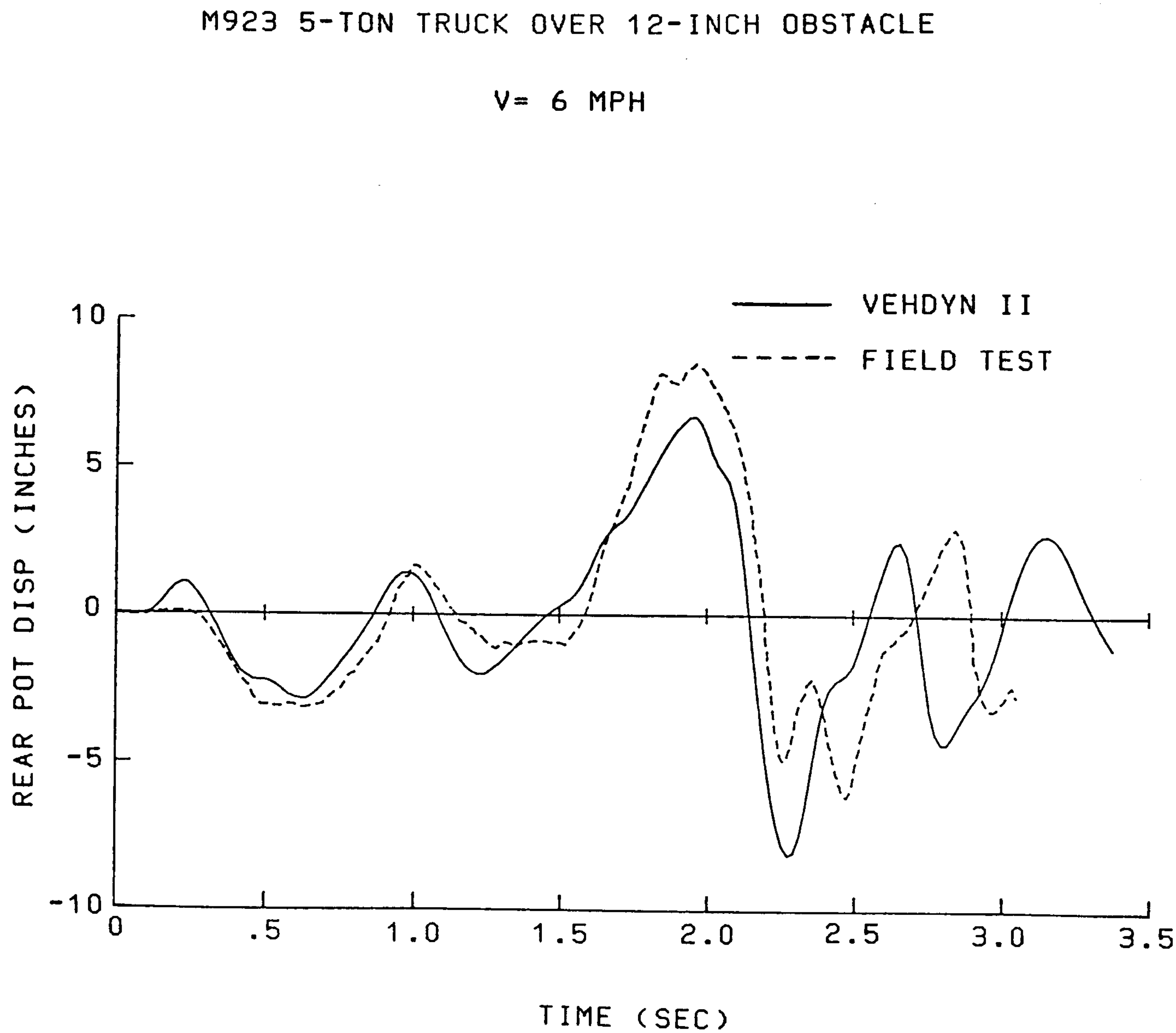
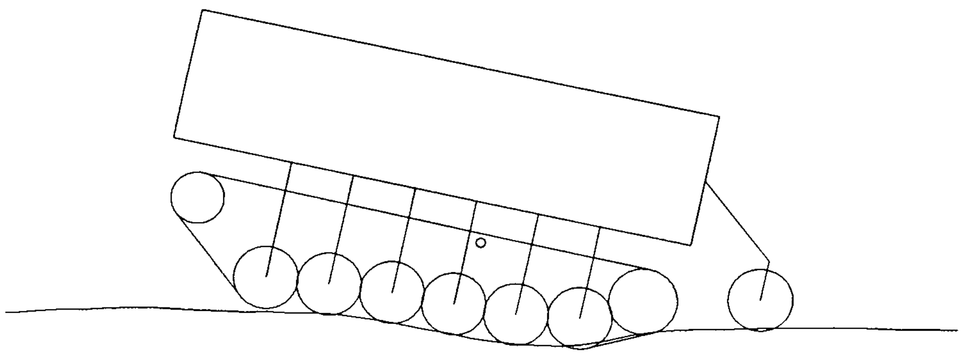


Figure 16. Comparison of measured and calculated rear yo-yo displacements for the M923 5-ton truck negotiating a 12-inch half-round obstacle at 6 mph.

M60 BATTLE TANK WITH MINE ROLLER / 24–IN. CLEARANCE (11/28/88)
LET TEST COURSE 7 LEFT TRACK 10–27–87
V= 8.00 MPH, T=16.40 SEC.



M60 TANK WITH STIFF DAMPERS AND MINE ROLLER (24–IN. CLEAR.) (11/28/88)
LET TEST COURSE 7 LEFT TRACK 10–27–87
V= 8.00 MPH, T=16.40 SEC.

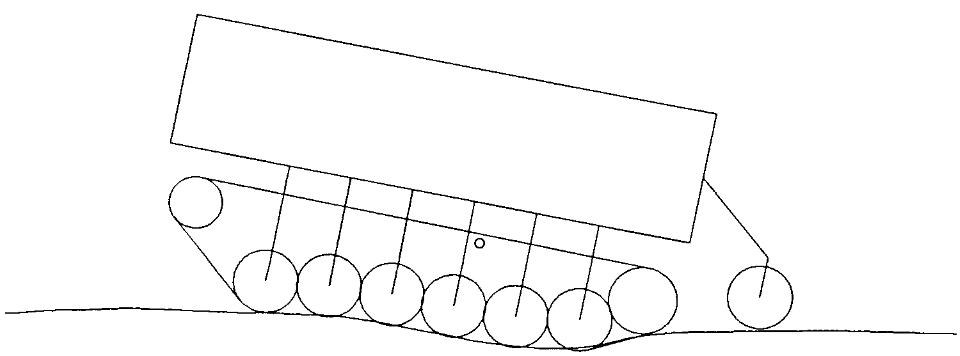


Figure 17. VEHDYN II simulation comparison for M60 mine roller with stock and stiffened suspensions.